Canadian Institute of
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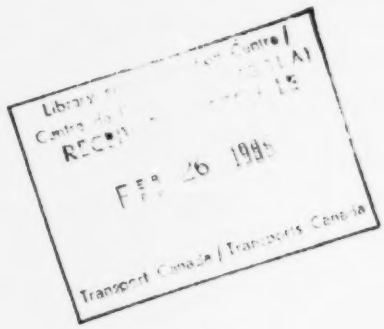

A COMPUTER SIMULATION ANALYSIS OF ELECTRIC LOCOMOTIVE SWAY

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17. Abstract

Pantograph lateral displacement from track centreline is investigated by means of computer simulation. The British Columbia Railway's electric locomotives (GF6C) are modeled in a 21-degrees-of-freedom computer model. The locomotive wheels are forced to follow the deviations in surface and cross-level recorded for a 90 mph (145 km/h) track. The same data are scaled according to track maintenance specifications so that pantograph displacements for a range of track conditions could be inferred. The influence of lateral wind loads is studied. The amplitude of motion increases as track quality decreases. The dependence of amplitude on speed is not monotonic. At intermediate speeds (30 km/h to 45 km/h) there is a coincidence of rail joint encounter frequency and body-roll natural frequency which leads to high amplitudes. The predicted motions of the pantograph are significantly less than those used in current design practice.

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17. Résumé

Les oscillations du pantographe de part et d'autre de l'axe de la voie ont été étudiées par simulation assistée par ordinateur, utilisant un modèle à 21 degrés de liberté élaboré exprès pour les locomotives électriques GF6C de la BC Rail. Ce modèle reproduit le comportement des roues d'une locomotive roulant à 90 mi/h (145 km/h) sur une voie dont on connaît les défauts de nivellement longitudinaux et transversaux. Il permet aussi de faire varier les paramètres de qualité de la voie, de sorte qu'on peut prévoir les oscillations du pantographe pour toute une gamme de niveaux d'entretien des voies. Les charges dues aux vents traversiers ont également été étudiées. L'amplitude des oscillations est fonction inverse de la qualité de la voie La relation amplitude-vitesses n'est pas linéaire. Aux vitesses intermédiaires, c'est-à-dire entre 30 et 45 km/h, la fréquence de franchissement des joints de rails et la fréquence propre de roulis du véhicule coincident, amplifiant les oscillations. Les oscillations prévues par simulation sont sensiblement inférieures à celles prises en compte couramment par les bureaux d'études.

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1. INTRODUCTION

This investigation was initiated by the Transportation Development Centre on the recommendation of the Standards Subcommittee of the Railway Electrification Committee (REC).

The REC was formed in October 1981 to assess electrification in the Canadian context. It includes representation from railways, various federal government departments, several provincial governments, the Railway Advisory Committee (RAC) and the Canadian Institute of Guided Ground Transport.

The objectives of the Standards Subcommittee are:

- to provide documented findings and tentative recommendations on standards which can be applied to railway electrification in Canada
- to produce a draft Canadian standards document on railway electrification.

The focus of the Standards Subcommittee has been those design standards which would have a major impact on the capital costs of electrification and which require early definition of national standards.

The cost of an overhead contact system is quite sensitive to the "blow-off" criteria (the lateral displacement of the contact wire from the pantograph). Blow-off results from the combined effects of contact wire deflection from wind forces and pantograph displacement due to locomotive sway. Existing design literature either assumes full locomotive body rotation to the suspension stops or some factor thereof.

1.1 Existing Design Practices

The lateral displacement of the pantograph from track centreline includes deviations of the rails from centreline as well as a range of locomotive factors which influence its lateral translational and rotational system. Figure 1 illustrates the locomotive elements involved. These elements, as well as various recommended allowances, are shown in Table 1.

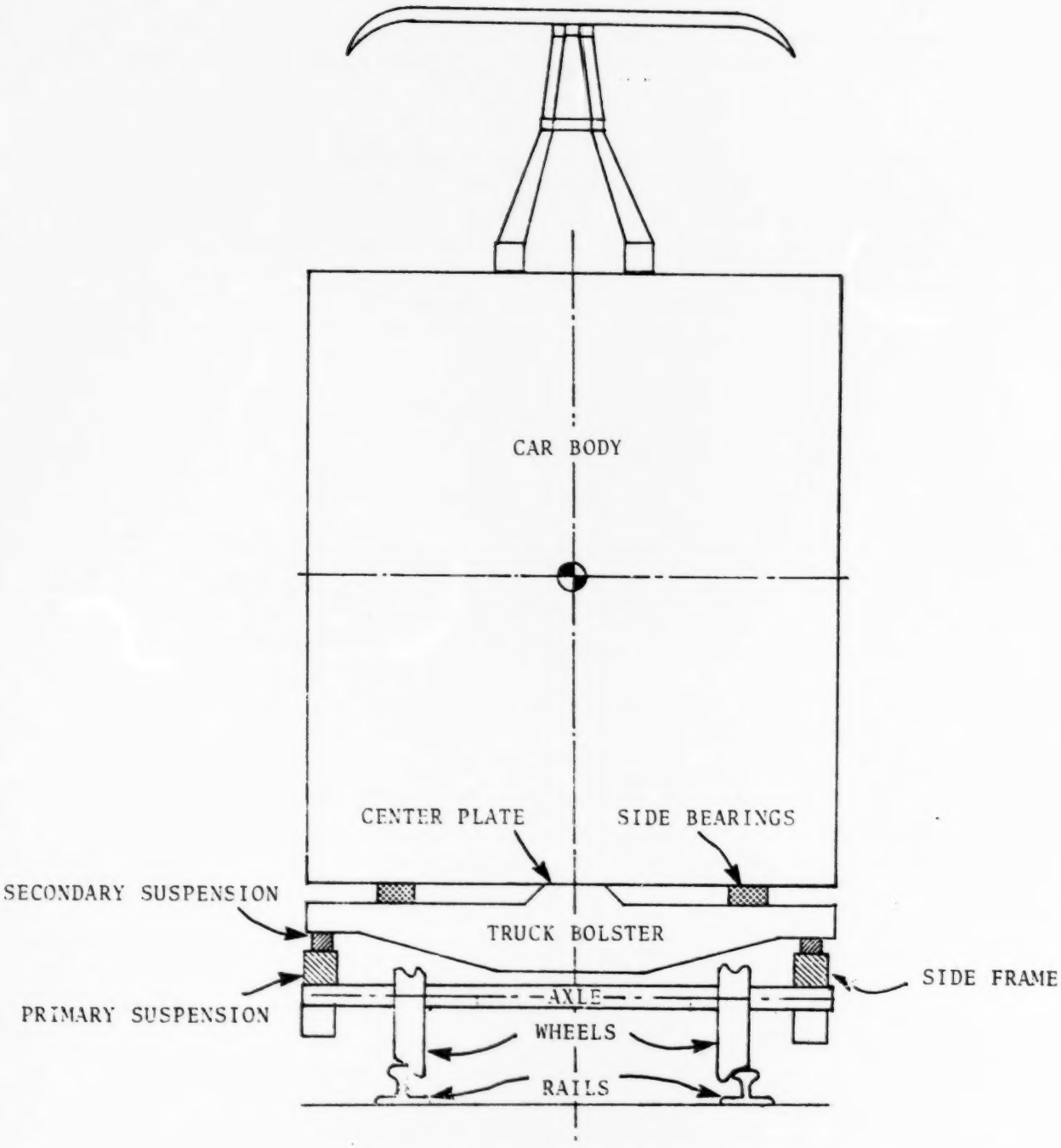


Figure 1 Components in locomotive sway

TABLE 1

LOCOMOTIVE SWAY AND CONTACT WIRE BLOW-OFF ASSUMPTIONS

	Displacement* (mm)			
	AREA Class 6 Track		N.E. Corridor	
	Locomotive	MU Car	MU Car	BCR
Locomotive and track:				
1. lateral	25.4	50.8	50.0	17.8
2. side bearing tilt	47.5	352.6	†	32.2
3. spring deflection	114.3		†	76.3
4. bolster swing	-		†	-
5. rails cross-level	56.9	56.9	†	79.8
6. pantograph sway	76.2	76.2	†	46.2
Subtotal (a)	320.3	535.2	370.00	252.3
7. unbalanced super-elevation on curves	-	-	356.11	-
Contact wire:				
8. erection tolerance	n/a	n/a	25.4	35.0
9. stagger effect	n/a	n/a	25.4	77.7
10. blow-off	n/a	n/a	237.2	252.0
11. pole deflection	n/a	n/a	50.8	-
12. temperature movement	n/a	n/a	66.0	-
Subtotal (b)	n/a	n/a	404.8	364.7
Total allowance ((a) + (b))	n/a	n/a	774.8	617.0

* Safety factors have been incorporated into the data as specified: AREA - none, N.E. Corridor - 0.5 of body roll, BCR - 0.7 of all factors.

† Details not provided.

While items 2, 3 and 4 in Table 1 are shown as static displacements, they might be interpreted as peak dynamic reaction to track irregularities. The values for the total pantograph displacement, based on the same factors, range from a low of 252.3 mm for the BCR to a high of 535.2 for the AREA recommendations. The AREA recommendations are for high-quality track. If the tolerances of the BCR Tumbler Ridge Branch Line were applied, the total AREA locomotive allowance would increase to 649.5 mm. With variations of accepted design standards of this magnitude and very little quantitative rationale provided, it is important to obtain a better understanding of the assumed relationships.

1.2 Objectives of this Study

The objective of this study is to improve the level of confidence associated with the "locomotive sway" design constraint. The method of achieving this is with computer simulations of a specific locomotive at various speeds over a range of track conditions.

2. THE COMPUTER MODEL

The mathematical model of the locomotive has been designed to satisfy the simultaneous requirements of providing accurate predictions and minimizing the complexity of the model. The purpose of the model is to predict the lateral motions of the top of the pantograph as the vehicle moves along tangent track with prescribed vertical and cross-level profiles.

When this goal is considered, it becomes clear that there are motions of the bodies comprising the vehicle which make negligible contributions to the lateral motion. These motions are in the longtudinal, pitch and yaw directions. Although significant in many other possible studies, they are ignored here because of their lack of involvement in the lateral motions of the pantograph.

2.1 The Vehicle

The vehicle is visualized as being composed of a total of nine rigid bodies, interconnected by elastic and damping elements. The bodies are: six axle/wheelsets, two truck frames, and one carbody. The

motions of these nine bodies are specified with 21 degrees-of-freedom, including vertical and roll motions of each component plus lateral motions of the truck frames and the carbody. An exploded view of the system is presented as Figure 2.

Each axle is connected at two points to a truck frame by the primary suspension elements. These connections are characterized by stiffness and damping in the longitudinal, lateral and vertical directions. There is a clearance limit in the lateral direction at the primary suspension points, after which no further lateral excursion is allowed.

There is an additional level of suspension where the truck frames are connected to the carbody. There are two such connections per truck, each modelled with tri-axial stiffness and damping. A lateral clearance limit exists at these points as well.

Modelling this complex vehicle as having relatively few components has required that assumptions be made concerning the placement of the mass properties which are not inherently part of the rigid bodies defined here. The traction motors, for instance, are essentially unsprung and have had their mass lumped on the axles. The bolsters have been included as part of the carbody.

The inertia and suspension properties are listed in Tables 2, 3 and 4. The data in these tables corresponds very closely to those given by Garivaltis and Garg.[1]

2.2 The Pantograph

The pantograph is located atop the locomotive and is designed to be flexible in the vertical direction so that it can maintain contact with the overhead wires as their position with respect to the locomotive changes vertically at different positions along the track. This study is, however, concerned with lateral, not vertical, motions of the pantograph. All details available about the device indicate that it is very stiff in the lateral direction.

No separate dynamic model of the pantograph structure has been derived for the work reported here. Any such model would be unrealistic and would not add to the accuracy of the predictions being

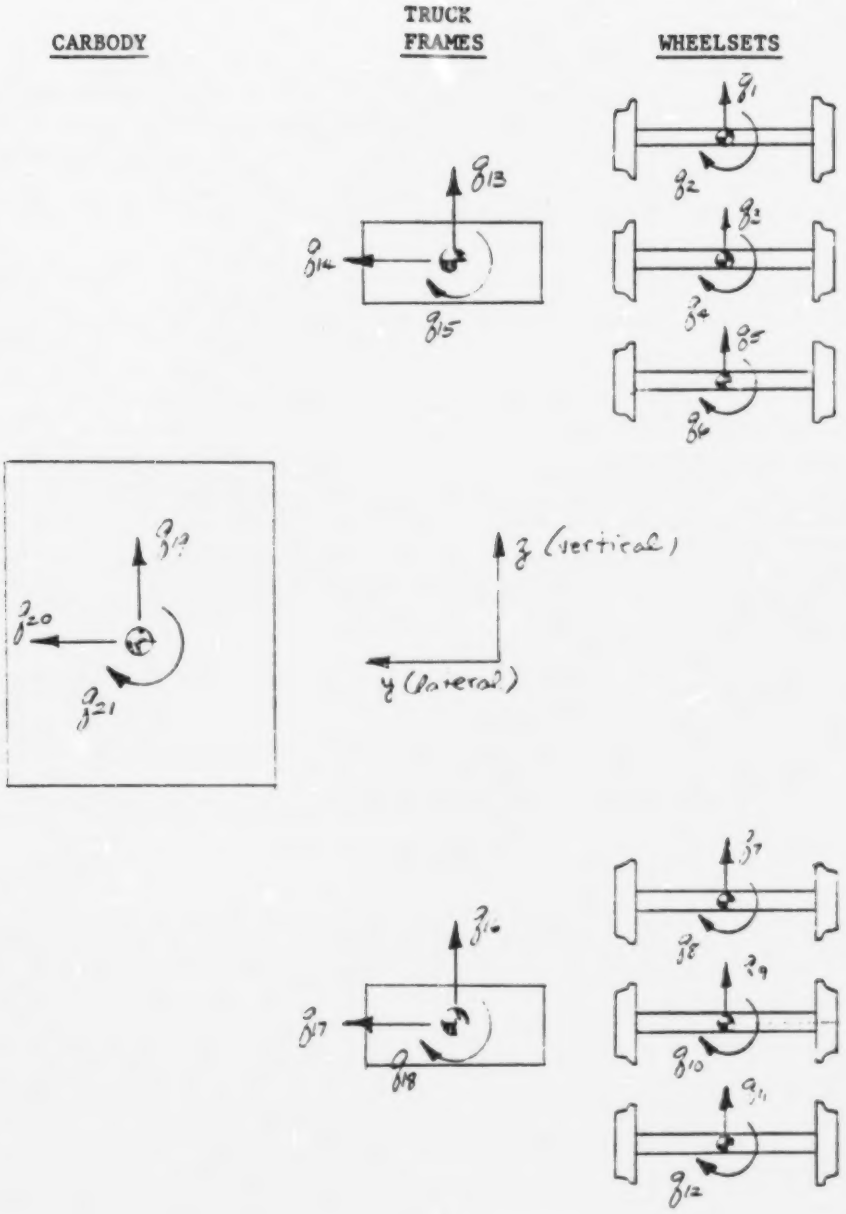


Figure 2 Exploded rear view of the system showing the degrees-of-freedom, q_i

TABLE 2

MASS PROPERTIES

Component	Mass (kg)	Roll Moment of Inertia ($kg\text{-}m^2$)	Position of c.g. with respect to Carbody c.g. (m)		
			x	y	z
Carbody	1.344×10^5	1.947×10^5	0	0	0
Leading truck	7018	633.9	7.014	0	-1.425
Trailing truck	7018	633.9	-7.014	0	-1.425
Wheelset #1*	5264	905.6	9.027	0	-1.598
Wheelset #2	5264	905.6	6.979	0	-1.598
Wheelset #3	5264	905.6	4.851	0	-1.598
Wheelset #4	5264	905.6	-4.851	0	-1.598
Wheelset #5	5264	905.6	-6.979	0	-1.598
Wheelset #6	5264	905.6	-9.027	0	-1.598

* The wheelsets are numbered rearward starting at the leading wheelset.

TABLE 3

SUSPENSION PROPERTIES

Suspension Level	Stiffness (N/m)			Damping (N.sec/m)		
	x	y	z	x	y	z
Primary	4.38×10^7	2.15×10^5	1.26×10^6	1.09×10^3	1.21×10^4	9.52×10^3
Secondary	1.226×10^6	1.226×10^6	2.627×10^7	5.57×10^4	5.57×10^4	4.38×10^4

TABLE 4

SUSPENSION PLACEMENT

Suspension Level	Position with respect to carbody c.g. (m) x	y	z	Notes
Secondary	±7.01	±0.892	-1.275	at four locations
Primary	±9.027	±1.003	-2.614	wheelsets 1 and 6
Primary	±6.979	±1.003	-2.614	wheelsets 2 and 5
Primary	±4.851	±1.003	-2.614	wheelsets 3 and 4

made. The primary action of the pantograph, in the lateral direction, is to amplify the effects of carbody roll by virtue of its extension above the carbody. This effect has been included in the analysis, with the top of the pantograph specified to be 6.86 m above the top of rail.

Motions resulting from lateral deflections of the pantograph itself will be small and can be superimposed on the predicted motions. The manufacturer's specifications show a lateral motion of less than 30 mm under a load of 500 N.

2.3 Wind Forces

The effect of lateral wind loads on the motion of the locomotive is modelled here as resulting from a steady lateral wind. The vehicle is modelled as a rectangular block having a drag coefficient, $C_D = 0.60$. The resulting wind loads act at the geometric centre of the locomotive and are calculated from:

$$F_w = C_D \left(\frac{1}{2} \rho A V^2 \right) \qquad (1)$$

where: F_w = steady lateral force

ρ = mass density of air

A = cross-sectional area of the vehicle (as seen when viewed from the side)

V = wind speed.

For the vehicle used here, the parameters are:

$$\rho = 1.235 \text{ N} \cdot \text{sec}^2/\text{m}^4$$

$$A = 90.6 \text{ m}^2$$

As a result, the lateral load due to a wind of 100 km/h (27.8 m/sec) would be:

$$F_w = (0.60)\left(\frac{1}{2}\right)(1.235)(90.6)(27.8)^2 = 2.6 \times 10^4 \text{ N}$$

This force is only slightly greater than 1 per cent of the total vehicle weight.

2.4 The Equations of Motion

The equations of motion for the system have been derived numerically using a technique developed by Anderson.[2] The governing equations are formed directly from an input file describing the system parameters such as listed in Tables 2, 3 and 4. As a result, the equations are not available in analytical form, but are numerical in nature.

This direct numerical evaluation, without requiring analytical effort on the part of the user, allows more effort to be placed on the solution of the equations and reduces the probability of making an error during the derivation.

The equations of motion have the standard form for linear, mechanical systems:

$$[M]\,(\ddot{x}) + [C]\,(\dot{x}) + [K]\,(x) = (F) \qquad (2)$$

where [M], [C] and [K] are square coefficient matrices. These matrices are derived numerically by the computer program described in reference [2].

The vector (x) contains the system degrees-of-freedom and is constructed so that it may be easily partitioned into two different types of freedoms.

There are those representing wheelset motions which are constrained to follow the undulations of the track. These twelve prescribed freedoms may be written as a vector (x_w). The remaining degrees-of-freedom are associated with the trucks and carbody and are not constrained except through the action of suspension elements. These can be regarded as another vector, (x_v), representing vehicle motions.

The equations of motion can then be partitioned as follows:

$$\begin{bmatrix} M_{ww} & M_{wv} \\ M_{vw} & M_{vv} \end{bmatrix} \begin{pmatrix} \ddot{x}_w \\ \ddot{x}_v \end{pmatrix} + \begin{bmatrix} C_{ww} & C_{wv} \\ C_{vw} & C_{vv} \end{bmatrix} \begin{pmatrix} \dot{x}_w \\ \dot{x}_v \end{pmatrix} + \begin{bmatrix} K_{ww} & K_{wv} \\ K_{vw} & K_{vv} \end{bmatrix} \begin{pmatrix} x_w \\ x_v \end{pmatrix} = \begin{pmatrix} F_w \\ F_v \end{pmatrix} \qquad (3)$$

The vector of externally applied forces has also been partitioned to show (F_w), the external forces acting on the wheelsets, and (F_v), the external forces corresponding to vehicle degrees-of-freedom.

The equation shown as (3) consists of two separate sets of equations. The unknowns in this problem are (x_v) (and its derivatives) and the wheel/rail forces represented by (F_w). All other variables are known. (x_w) and its derivatives may be determined from the rail profile. (F_v) contains only the aerodynamic body force since the vehicle has no other external forces acting upon it. Taking all known quantities to the right-hand side of the equation of motion representing the vehicle gives:

$$[M_{vv}](\ddot{x}_v) + [C_{vv}](\dot{x}_v) + [K_{vv}](x_v) = (F_v) - [M_{vw}](\ddot{x}_w) - [C_{vw}](\dot{x}_w) - [K_{vw}](x_w) \qquad (4)$$

These equations are numerically integrated with respect to time to produce predictions of the vehicle motion, as characterized by (x_v), $(\dot{x}_v)$ and $(\ddot{x}_v)$. Once these quantities are known, they can be substituted into the remaining equations to find the wheel/rail forces:

$$(F_w) = [M_{wv}](\ddot{x}_v) + [C_{wv}](\dot{x}_v) + [K_{wv}](x_v) + [M_{ww}](\ddot{x}_w) + [C_{ww}](\dot{x}_w) + [K_{ww}](x_w) \quad (5)$$

For this study, the wheel/rail forces were not required and were not calculated.

2.5 Lateral Clearances

One aspect of the locomotive which has an important effect on pantograph motion is the limitation on lateral motion provided by bump stops. These stops are provided at both the primary and secondary suspension levels.

The model includes the primary bump stops in an approximate way by simply increasing the stiffness of the primary suspension to levels where the clearance limitation is not violated. The clearance at this level is small (approximately ±5 mm) so that errors introduced by the assumption of a stiff, linear suspension are not significant.

The clearance at the secondary suspension level is ±44.5 mm. Motions of this magnitude are important to the overall motion of the pantograph and must be properly taken into account in the vehicle model. The secondary suspension has therefore been modelled in a non-linear fashion. The normal, linear secondary suspension, which is relatively soft, is supplemented by a non-linear element which has zero stiffness within the allowable clearance limits and becomes very stiff for motions outside the limits.

The non-linear elements are included in the equations of motion as part of the applied force vector (F_v) of equations (3) and (4). Although these forces are not externally applied forces, they may be properly included here by accounting for their equal and opposite effects on adjoining bodies.

3. TRACK PROFILE

Accurate predictions of vehicular motion are as strongly dependent upon good modelling of the forcing input as they are upon good modelling of the vehicle. The locomotive studied here moves in response to variations in vertical and cross-level profiles as it moves along the track. These variations, when plotted, appear to behave as random variables (Figures 3 and 4). These figures show the variations found on a quarter-mile section of good-quality, jointed track near Kingston, Ontario. These data were originally in the form of a strip-chart recording.

In order to use the track data as input to the vehicle simulation, it had to be in digital form and had to be amenable to differentiation to allow ($\dot{x}_w$) and ($\ddot{x}_w$) to be determined (Section 2.4). The processing of the data involved a number of steps.

First, the strip chart traces were digitized using a Hewlett-Packard 9872T digital plotter with ±0.025 mm resolution to create a computer file of 501 data points for vertical profile and another 501 for the cross-level profile. These data are equally spaced along the track, corresponding to one data point every 0.80 m.

In order to satisfy the requirement for a differentiable function, these data were approximated by a Fourier series of the form:

$$f(x) = \sum_{i=1}^{N} a_i \cos [(i-1)(s_1 x - s_2)] + b_i \sin [(i-1)(s_1 x - s_2)] \quad (6)$$

where: x = the position along the track

$f(x)$ = vertical or cross-level profile as a function of position along the track

a_i, b_i = Fourier coefficients

s_1, s_2 = scaling factors which transform the range of x to $0 \leq x \leq 4\pi N/2N+1$

N = number of terms in the series.

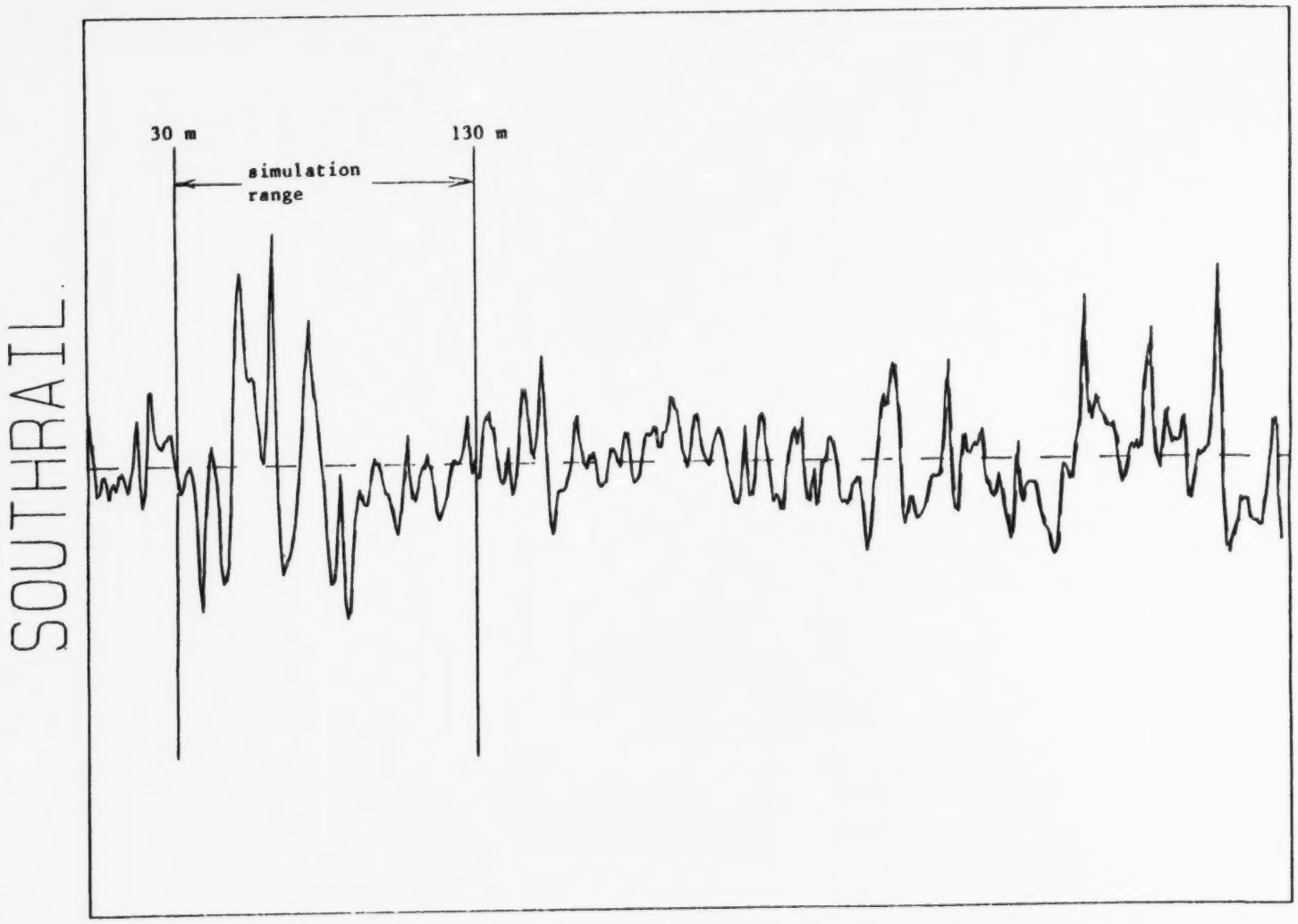


Figure 3 Vertical profile over quarter-mile section of jointed track

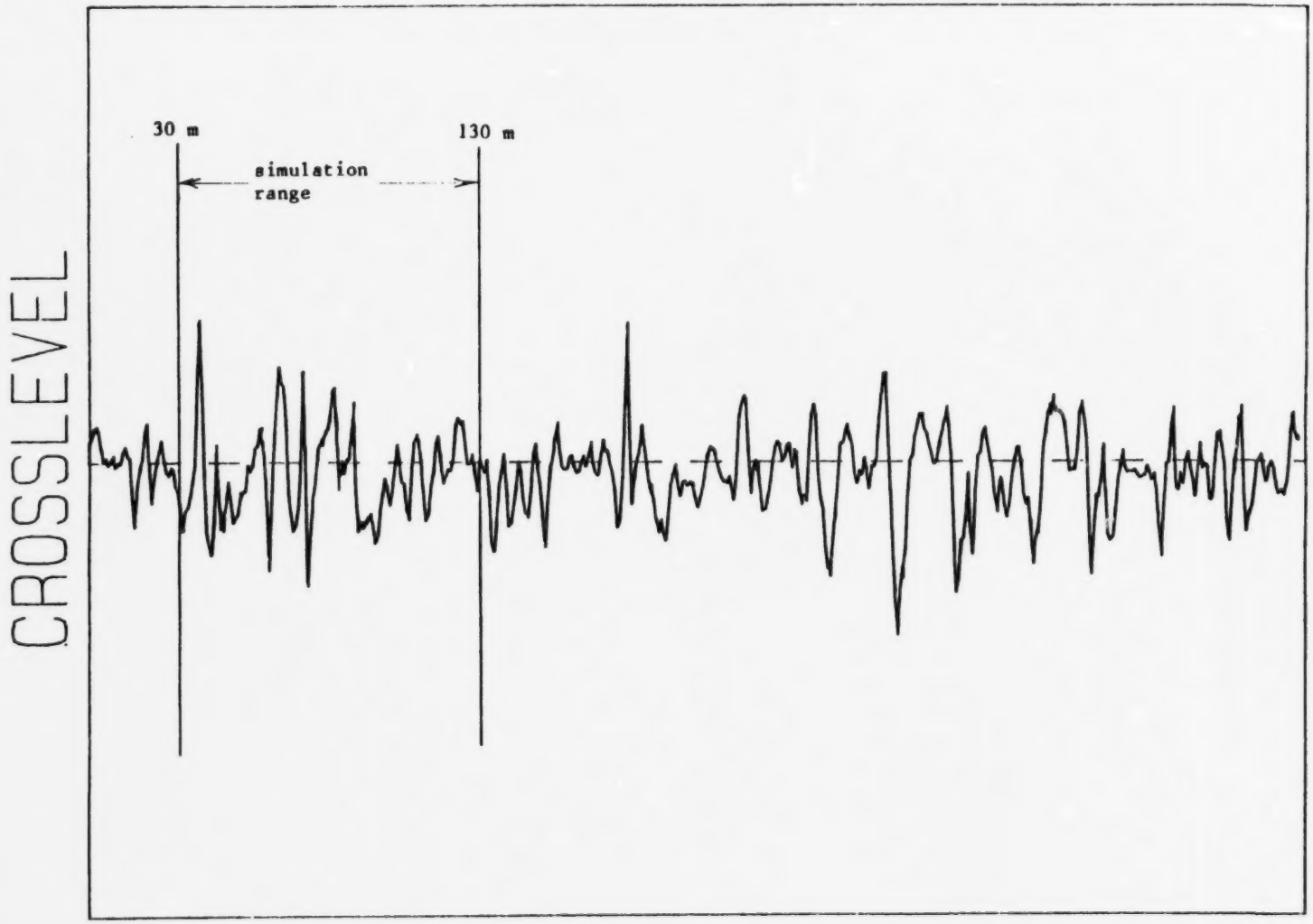


Figure 4 Cross-level profile over quarter-mile section of jointed track

For the cases considered here, N = 200 was used. The curves of Figures 3 and 4 actually contain both the measured values and the Fourier series calculated to represent them. The agreement is excellent everywhere.

The measured track profiles are used only as shape functions in the simulations. They have been normalized with respect to the absolute value of the largest peak values found in the samples. The range of the vertical and cross-level profiles used in the simulations is therefore -1.0 to +1.0. The data are scaled to any desired track tolerance simply by multiplying the normalized shape functions by the maximum deviations allowed in the maintenance standards for the class of track being considered. The values of Table 5 have been used.

TABLE 5

ALLOWABLE TOLERANCES
(mm)

Speed		Line	Surface	Variation in Cross-Level
km/h	(mph)			
145	90	10	13	25
140/135	85	10	13	25
130	80	13	13	25
125/120	75	13	13	25
115/110	70	13	16	29
105	65	16	16	29
100/95	60	16	16	32
90/85	55	16	19	32
80	50	19	19	35
75/70	45	19	19	35
65	40	19	22	38
60/55	35	22	25	41
50/45	30	25	25	44
40	25	29	29	48
35/30	20	32	32	51
25/20	15	35	35	64
15	10	38	38	76

Marked on Figures 3 and 4 is a simulation range which spans 100 m near the beginning of the quarter-mile section. This range was chosen for the following reasons:

- simulations over the entire track section were prohibitively expensive and did not produce more information than could be gathered from the 100 m section
- the section chosen contains a number of large-amplitude disturbances
- the initial track section (0 to 30 m) is used as a settling range so that results are not used where the vehicle has just left perfectly horizontal track and is entering the disturbed section of track. Simulations are actually run from 0 to 130 m, but results prior to 30 m are not used.

4. RESULTS

The cases simulated are summarized in Table 6. The detailed responses for the 145 km/h track cases are presented in Appendix A. In this section, some overall characteristics and trends determined from the time-domain simulations are presented.

TABLE 6

CASES RUN

Permissible Track Speed (km/h)	Cross-Wind Speed (km/h)	Locomotive Speeds (km/h)
145	0	15, 45, 105
145	100	15, 30, 45, 60, 75, 90, 105
100	100	15, 30, 45, 60, 75, 90, 100
70	100	15, 30, 45, 60, 70
55	100	15, 30, 45, 55
30	100	15, 30

4.1 Sensitivity to Speed

Figure 5 shows the maximum pantograph deviation versus speed for all of the track conditions considered. The values plotted here are the largest peak responses, as determined from the time traces, including the effect of a 100 km/h lateral wind.

All of the curves plotted in Figure 5 exhibit similar peaks at a speed of 30 km/h. The largest amplitudes are found at this speed, regardless of the quality of the track. This effect is caused by the regular roll disturbance of the staggered rail joints which are encountered every 5.94 m (19.5 ft). At a speed of 30 km/h, the encounter frequency is 1.40 Hz. An eigenvalue analysis of the vehicle was performed and showed that the natural frequency of the body in its roll mode is 1.88 Hz. The proximity of the natural frequency and the disturbing frequency lead to the high peak response at 30 km/h. At higher speeds, the disturbing frequency is above the natural frequency, resulting in smaller amplitudes of motion.

Figure 5 also shows the increased response as the track quality decreases. The worst case corresponds to 30 km/h on track where the permissible track speed is 30 km/h. The peak lateral motion of the pantograph is predicted to be 141 mm under this condition.

4.2 Probability Analysis

The predicted response of the pantograph is essentially random in nature. It corresponds to the output of a non-linear mechanical system being forced by a random track input. One way of characterizing the motion of the top of the pantograph is in terms of the probability of exceeding a specified displacement. Figures 6 and 7 show the probability of exceeding a displacement versus the displacement itself for two cases. The probabilities are calculated from the simulation results.

Figure 6 shows the probability of exceeding specified lateral displacements for six different speeds on 100 km/h track. The figure shows only the probabilities associated with deviations from the track centreline in the opposite direction to which the wire is blown. The fact that the highest probability seen here is 40 per cent is indicative

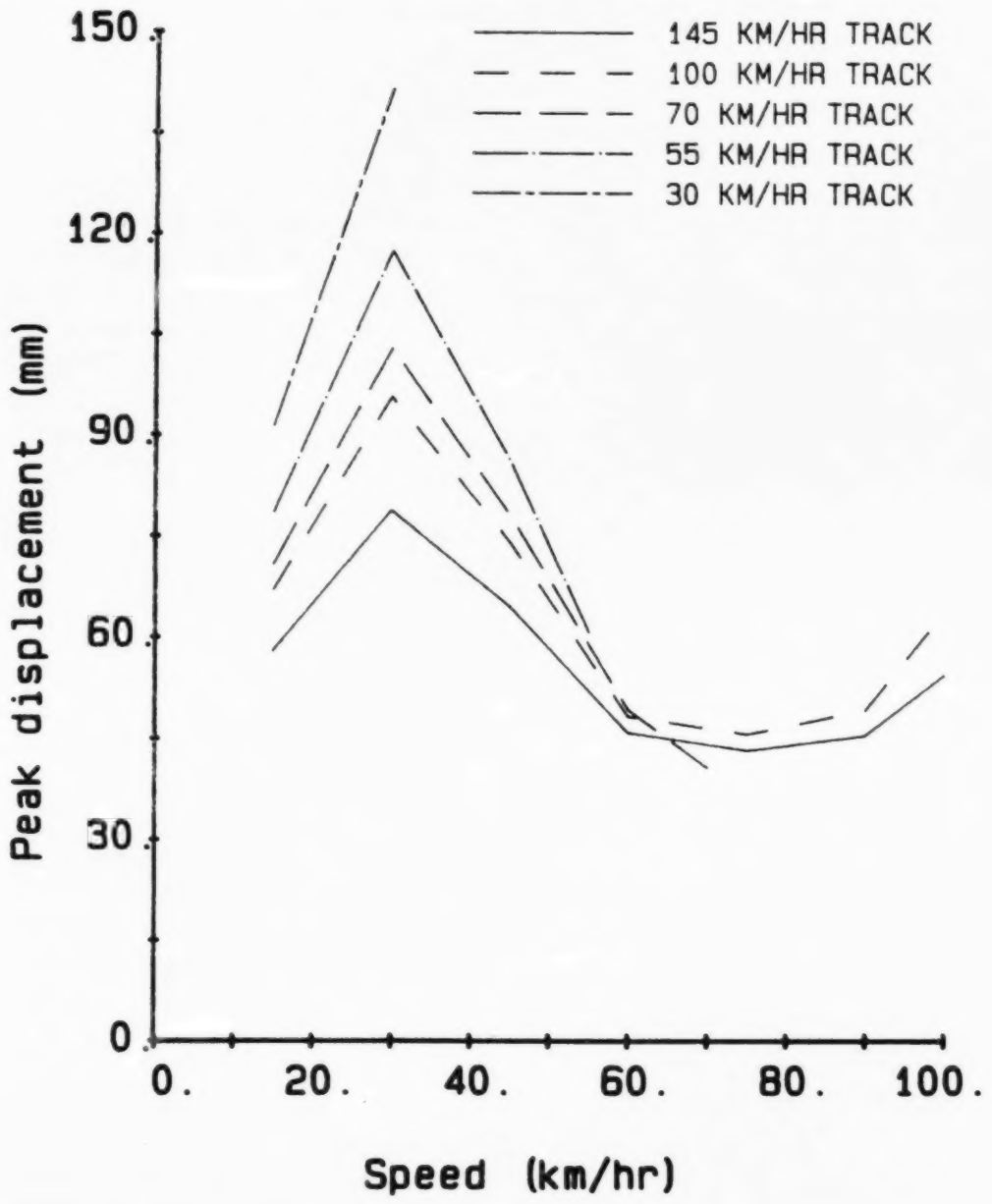


Figure 5 Maximum pantograph deviation versus speed

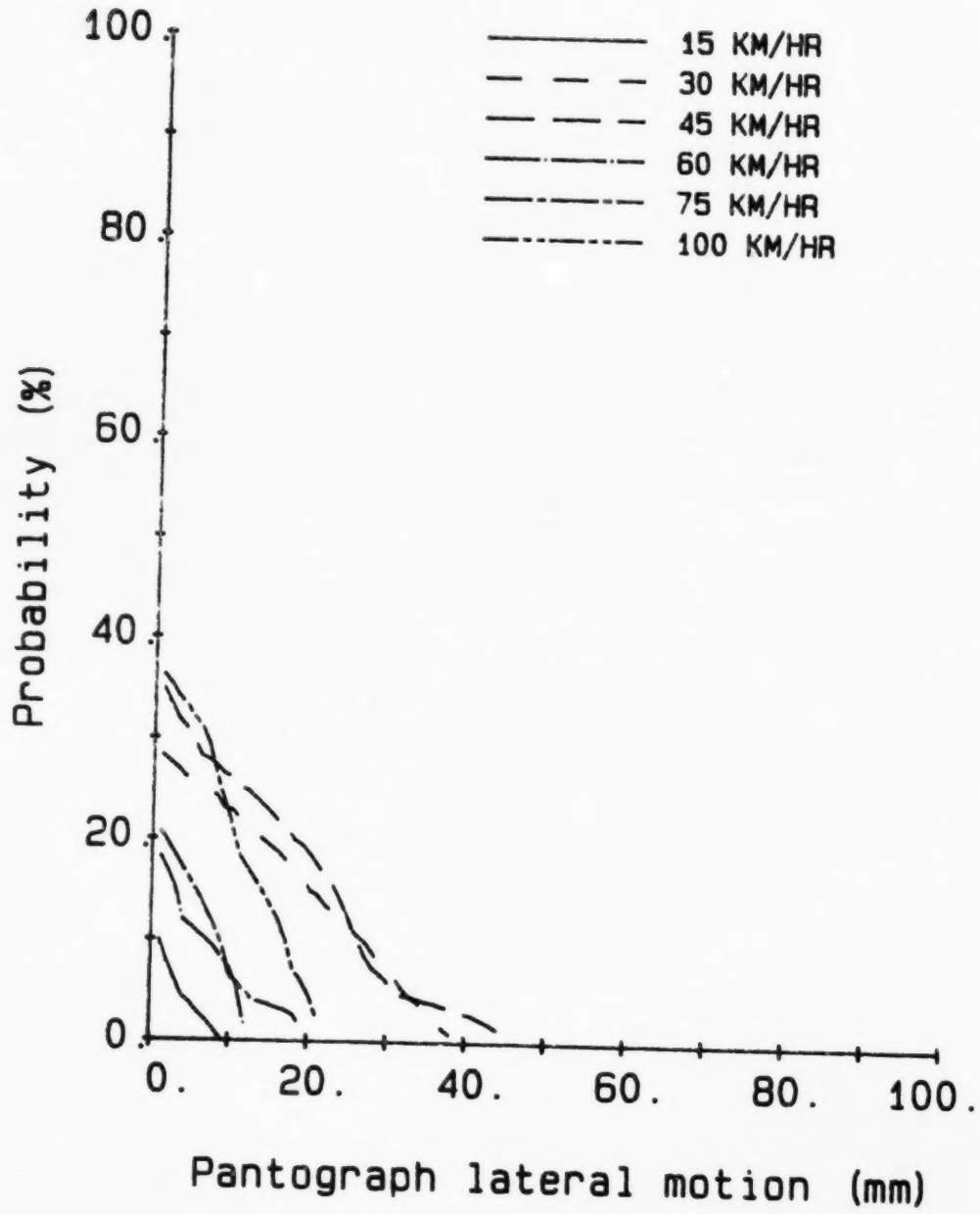


Figure 6 Probability of exceeding specified lateral displacements for six speeds on 100 km/h track

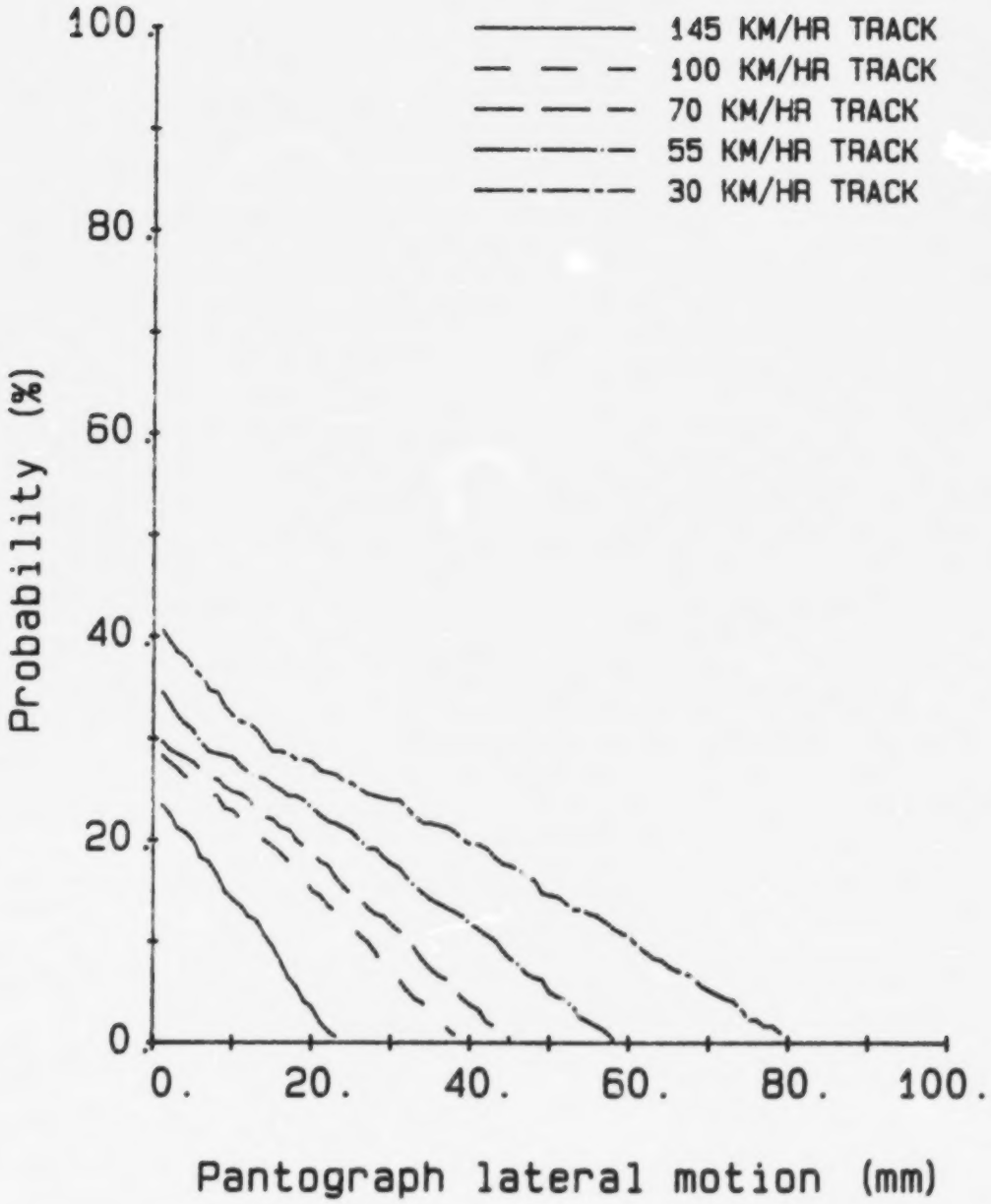


Figure 7 Probability of exceeding specified lateral displacements at 30 km/h on five different qualities of track

of the fact that the majority of the excursions will be in the same direction as the displaced contact wire. These displacements are measured with respect to the centreline of the track. The curves rapidly decrease with lateral displacement, with the probability of exceeding 35 mm being less than 5 per cent for all speeds. For the most part, a speed of 45 km/h provides the highest probability of exceeding a given displacement. The least probability of exceeding any given displacement is afforded by very low speeds (15 km/h) and by moderately high speeds (60 and 75 km/h). The probability increases further as the speed increases above 75 km/h.

Figure 7 shows the effect of track quality on the probability curves. All of the results are for a speed of 30 km/h where the highest peak motions are found. The curves show that the probability of exceeding a given displacement increases with decreasing track quality. It is apparent from these curves that the probability of exceeding 70 mm of motion is less than 5 per cent for any track quality, even at this severe speed.

4.3 The Effect of Wind

The effect of a steady lateral wind load caused by a 100 km/h wind is primarily to cause a shift of the mean values of the displacement. The figures in Appendix A show that the curves are shifted away from the zero displacement case in a positive sense. This is due to the assumed direction of the wind. Table 7 shows the mean values of the displacement for three simulations which were run with and without wind. The mean displacement without wind averages -6.6 mm and with wind averages +17.7 mm for the track considered. The mean displacement is therefore shifted by approximately 24 mm due to the 100 km/h lateral wind. This shift is in the same direction as the contact wire displacement and therefore reduces the net pantograph-contact wire misalignment.

Figure 8 shows the probability of exceeding specified displacements away from the contact wire, under conditions with and without the 100 km/h lateral wind. In the case of zero wind force, all motions are away from the wire so that probabilities go up to 100 per cent at small excursions. On the other hand, when there is wind, only those motions which move the pantograph away from the wire are included.

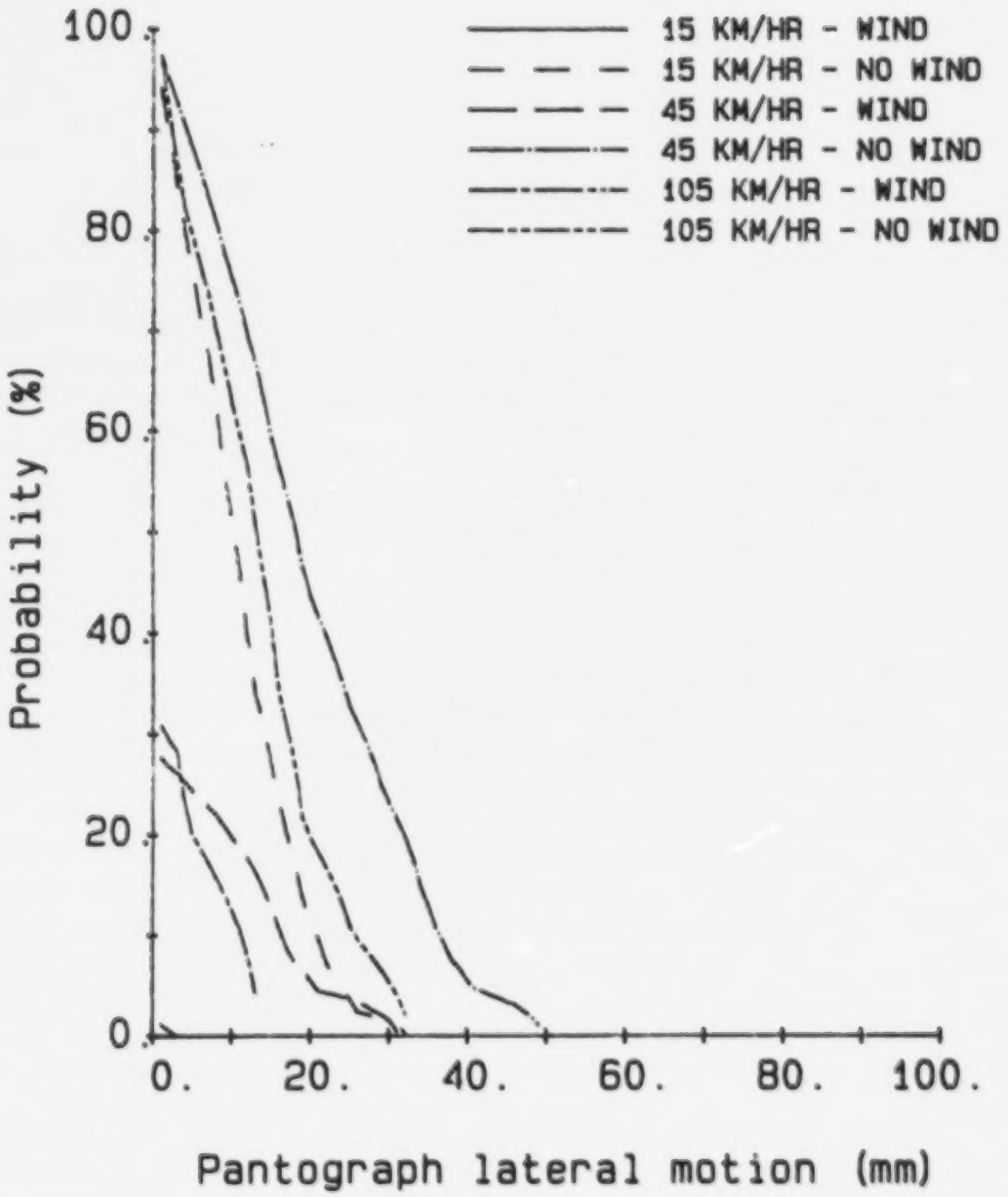


Figure 8 Effects of wind on the probability of exceeding lateral motions

TABLE 7

EFFECT OF WIND ON MEAN DISPLACEMENT

Forward Speed (km/h)	Mean Displacement (mm)	
	100 km/h Wind	No Wind
15	17.9	-6.9
45	17.9	-7.0
105	17.3	-6.0

5. CONCLUSIONS AND RECOMMENDATIONS

A computer simulation of the motions of a six-axle locomotive has been used to predict the lateral motions of the pantograph as the vehicle traverses jointed track, of different qualities, at varying speeds. The object of evaluating current design practice as regards "locomotive sway" assumptions included in the design of an overhead contact system for electrification has been accomplished. Some of the important findings of this study are discussed in this section.

It has been shown that the peak lateral displacements of the pantograph do not increase monotonically with speed. There is a speed where the regular spacing of rail joints disturbs the vehicle at a frequency close to the natural frequency, in roll, of the carbody. This speed results in the highest peak amplitudes. For the locomotive parameters used here, this speed was in the range of 30 km/h to 45 km/h.

When compared to the recommended values for total pantograph displacement used in existing design practice (Section 1.1), those predicted here are very small. The highest amplitude predicted here is 142 mm. This occurs in the leeward direction of a 100 km/h side wind and with track roughness typical of a 30 km/h track. The maximum amplitude in the windward direction for this case is only 80 mm. However, the model did not incorporate lateral shift in track, side-bearing tilt or pantograph slack. With these factors removed from

Table 1, the comparable locomotive sway presently cited for design purposes is in the range of 156 to 171 mm. In terms of probability, the likelihood of exceeding a total displacement of 100 mm is less than 5 per cent under any circumstances considered here. When the effect of a steady side wind is considered to act in the same direction as the shifted contact wire, the maximum sway is reduced even further.

The conclusion here must be that the existing design practices are overly restrictive and their applicability to future electrification in Canada should be reconsidered.

The effect of lateral wind loads on the pantograph motion has been established by simulating motions with and without a 100 km/h steady sidewind. A wind of this magnitude is predicted to shift the mean value of the pantograph motions by 24 mm in the direction of the wind. The displacement corresponding to lower wind speeds can be expected to decrease by the ratio of velocity squared due to the relationship of wind force to wind speed (equation (1)). The expected displacement due to a 50 km/h wind will therefore be one-quarter of that due to a 100 km/h wind.

The following areas of further work are suggested by the results reported: an evaluation of revisions of the locomotive suspension design, experimental validation of the predictions of the computer model, and more detailed simulation and field investigations of gusting side winds.

The peak pantograph amplitudes occur due to a resonance between rail joint inputs and body natural frequencies. A natural approach to dealing with these peaks would involve a modificaton to the secondary suspension of the locomotive. It is unlikely that the "critical speed" can be raised above the operating speeds without seriously degrading ride quality of the vehicle, leading to a harsh internal operating environment. It is feasible, however, to consider the introduction of non-linear (e.g. rubber) suspension elements which will absorb energy from the vehicle in the desired speed range without significantly affecting ride quality. The trade-offs to be considered when contemplating any suspension changes must include the benefits of reduced roll and catenary cost savings as opposed to the disadvantages of

increased track forces and maintenance costs. This type of study is best accomplished using computer models.

The dramatic effect of a steady lateral side wind on the peak pantograph displacement in relation to the contact wire has significant design and cost implications. However, this has been a preliminary evaluation and does not provide enough confidence to warrant major changes to the blowoff design criteria. Further simulation investigation of gusting winds and field observations would be desirable.

As the evaluation has been based on a simulation model, it would be desirable to validate the model predictions with field experience. As the BCR Tumbler Ridge Branch Line locomotives have been modeled, it would be desirable to validate the model there. This would involve three principal tasks:

- measurement of track profiles, including crosslevel and alignment deviations for a track section
- measurements of locomotive response in traversing the test track segment for a range of speeds
- comparison of detailed measurement with detailed simulation predictions.

REFERENCES

1. D.S. Garivaltis and V.K. Garg, "The Response of a Six-Axle Locomotive to Random Track Input," Report No. R-312, Association of American Railroads, Research and Test Department, June 1978.

2. Anderson, R.J., "A General, Multi-Body Approach to Dynamic Modelling of Rail Vehicles," Proceedings of the 1983 ASME Computers in Engineering Conference, Vol. 1, pp. 223-228.

APPENDIX A

PANTOGRAPH MOTION ON A SIX-AXLE LOCOMOTIVE

The predicted response of the locomotive for six different speeds and one track condition are presented in this appendix. The speeds cover the range from 15 to 105 km/h in 15 km/h increments. The track profiles are scaled to represent the track maintenance standards for 75 to 90 mph operation (13 mm vertical; 25 mm cross-level). In each case there is a 100 km/h lateral wind acting on the vehicle. The shift in the leeward direction is evident in each case. Figure A.3 superimposes the no-wind displacement to accentuate this shift. The predicted lateral motion of the pantograph as a function of position of the leading axle of the locomotive within the simulation section of the track is presented.

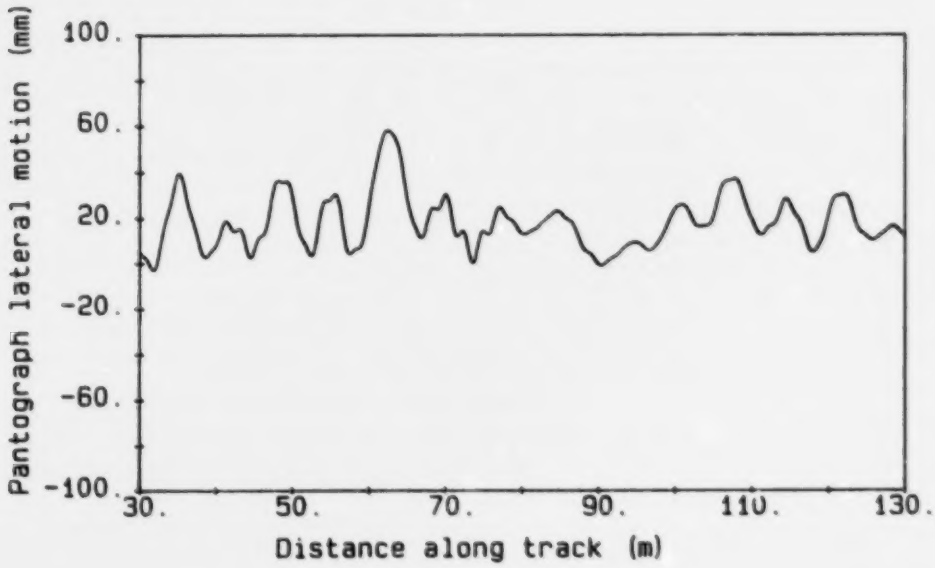


Figure A.1 Pantograph lateral motion at 15 km/h

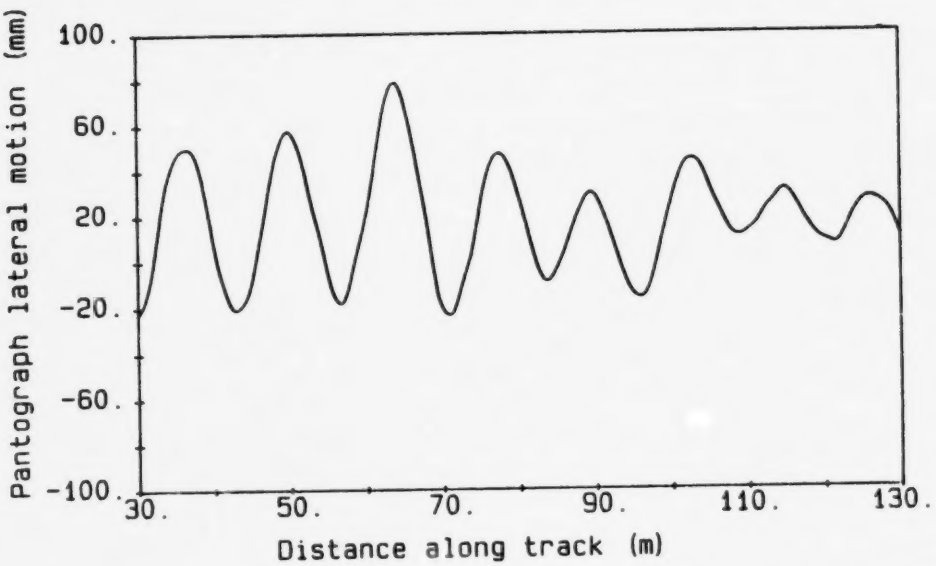
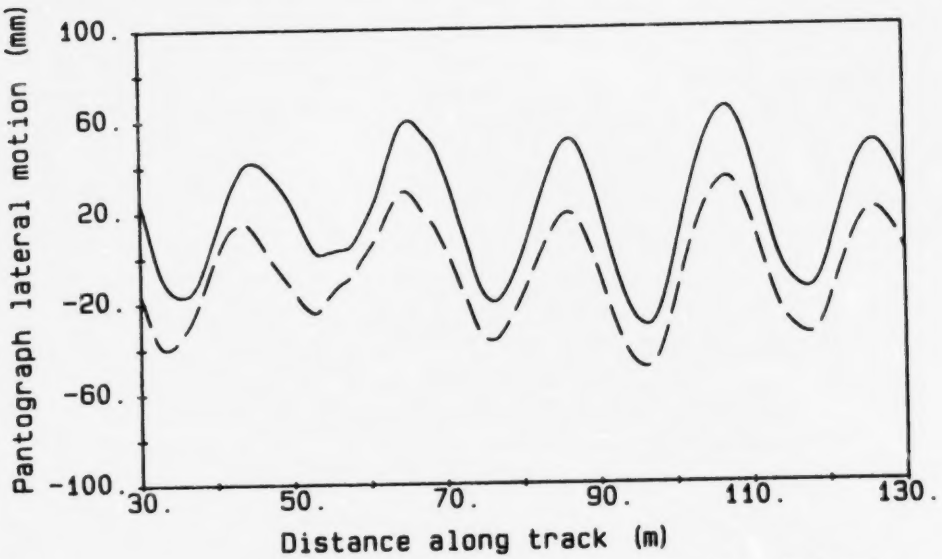


Figure A.2 Pantograph lateral motion at 30 km/h



dashed line -- no wind, mean value -7.0 mm
solid line -- 100 km/h wind, mean value +17.9 mm

Figure A.3 Effect of wind on pantograph response at 45 km/h on high-quality track. Dashed line

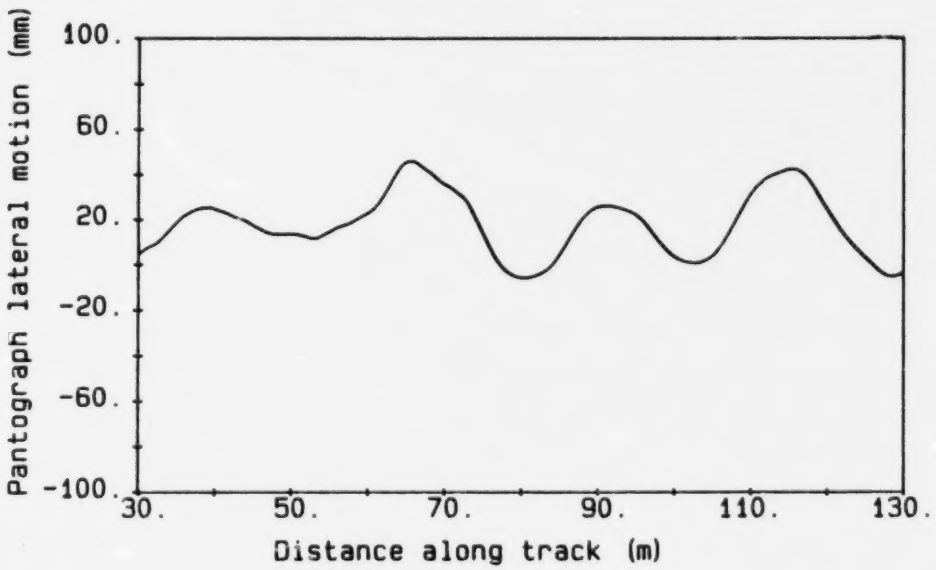


Figure A.4 Pantograph lateral motion at 60 km/h

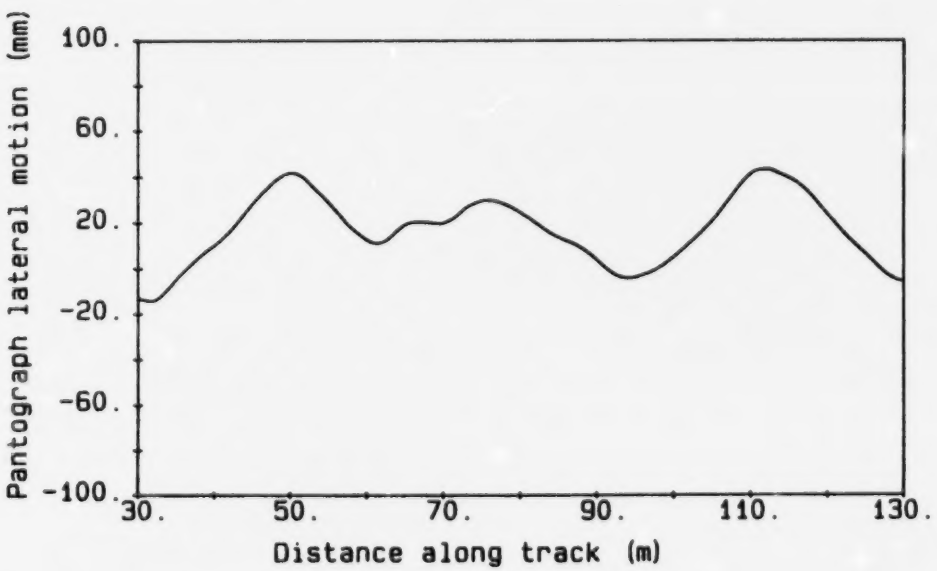


Figure A.5 Pantograph lateral motion at 75 km/h

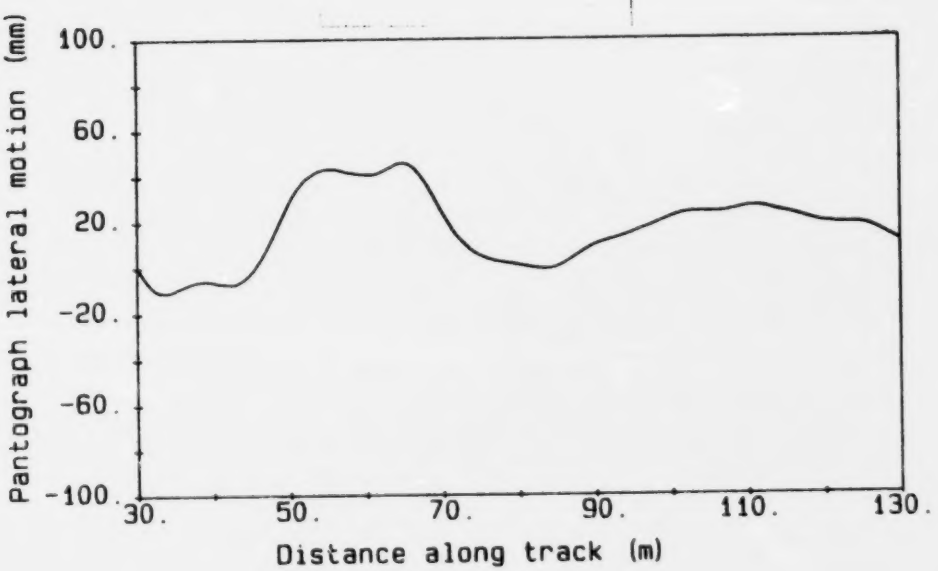


Figure A.6 Pantograph lateral motion at 90 km/h

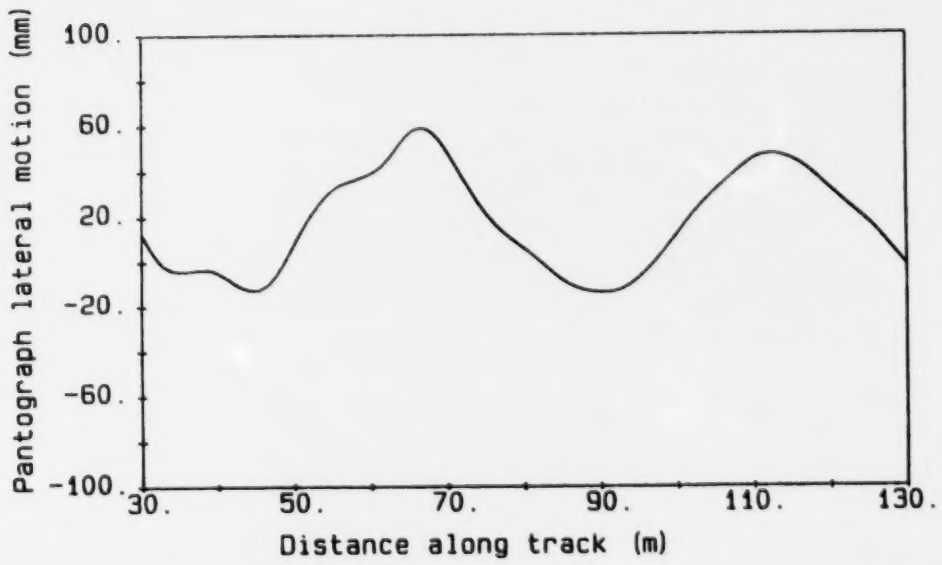


Figure A.7 Pantograph lateral motion at 105 km/h